Energy Analysis of Selected Air Distribution System of Heating, Ventilation and Air Conditioning System: A Case Study of a Pharmaceutical Company

DILEEP KUMAR*, RIZWAN AHMED MEMON**, AND ABDUL GHAFOOR MEMON**

ABSTRACT

The higher energy consumption causes environmental degradation along with depletion of conventional energy resources. The share of energy consumption in buildings is increasing with urbanization and that ultimately requires effective measures for energy conservation. In buildings, HVAC (Heating Ventilation and Air Conditioning) systems require huge amount of energy. This paper estimates the effects of compression of duct insulation of an HVAC system onthe auxiliary power consumption and temperature of supplied air. A mathematical model is developed in EES (Engineering Equation Solver) to ascertain these effects. The simulation results show that the cooling loss due to the insulation compression is about 14%. By increasing the insulation thickness from 10-40mm at selected points, the heat gain is estimated to decrease from 4.29-2.46kW. In addition to that effects of compression of thermal insulation on GHG (Greenhouse Gas) emission are investigated to reduce from 4.2-2.3kg/kW. Subsequently, the AC (Auxiliary Consumption) and temperature of the supplied air decrease by 5% and 0.4ºC, respectively.

Key Words: Heating Ventilation and Air Conditioning System, Energy Conservation, Thermal Insulation, Heat Gain.

1. INTRODUCTION

Rapid growth in industrialization and population has increased the energy consumption in various sectors throughout the world [1]. The building services are an energy intensive sector. The main service in a building application is HVAC system, which is designed to maintain and control the space condition to a comfortable and healthy indoor environment [2]. It is stated that around 40% of the US total energy consumption is in buildings, of which 60% is utilized in the HAVC systems [3,4]. It contributes as 40% of total CO_2 emissions in all energy sectors [5,6]. The inefficiencies in an HVAC system are mainly caused by heat gain in the air distribution ducts due to improper insulation design and other control parameters [7]. The optimal design of an HVAC system with specific consideration in the duct design for supply air can conserve a significant amount of energy [3]. In commercial and industrial sectors, there is a vast application of HVAC systems for building services

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and industrial processes. Therefore, optimal design of duct is crucial for energy savings in these sectors which ultimately led to a lower environmental impact and energy costs. Numerous researchers have conducted energy and cost based analyses of different duct systems [8-10]. The energy consumption data of building shows that there is a great potential of energy savings through effective thermal protection. The effectiveness of thermal performance of an HVAC system reduces the fuel consumption and environmental pollution as well [9]. The economical and optimal thickness of insulation is a function of design, operation and economic parameters of air distribution system [11]. The economical and optimum insulation thickness considers the initial cost incurred on the insulation plus cost saved in terms of energy savings during the expected lifetime of system [12].

The concept of degree-time is mostly used to calculate the thickness of thermal insulation along with its material. It is considered as simplest method mostly employed under static conditions. The economic analysis is carried out to determine the insulation thickness for different piping network used to transport oil in an industry. Their analysis was based on non-linear cost functions i.e. initial cost of material and annual energy savings. Rockwool and calcium silicate was used as insulation material, nominal size of piping was chosen between 0.1-0.273mm and working fluid was superheated steam, crude oil and 300 different distillates were considered along with constant convective heat transfer coefficient [13].The computer code is generated by Oztürk, and Karabay [14] to determine the optimum insulation thickness and size of the pipe. The developed computer code uses thermo-economic parameters to determine exergy destruction of piping network. The control theory approach and steepest descent method was used to determine the optimum insulation thickness for a circular pipe. In this analysis, thermal conductivity of material was constant, whereas bulk fluid temperature inside the pipe and surrounding

was varied [15]. Soponpongpipat et. al. [16] estimated optimum insulation thickness wrapped on HVAC duct varying convective heat transfer coefficient inside and outside the duct. In this analysis galvanized steel duct of 0.5m diameter with two different insulation material i.e. rubber and fiberglass were selected. From obtained results it is concluded the optimum thickness of insulation is independent of convective heat transfer coefficient of duct but energy savings is susceptible to it. Kecebas et. al. [17] calculated optimum insulation thickness, energy savings for a period of 10 years and payback period of five different pipe sections using four different energy sources in the city of Afyonkarahisar, Turkey.

HVAC system equipment such as pump, fan and compressors are known as auxiliary equipment. Although in most engineering analysis, AC of HVAC system is overlooked. In commercial buildings both fan and pump accounts for around 25% of its total energy consumption [18].Korolija et. al. [20] presented AC of different HVAC system types used in office building. It was noted that all air systems have higher values of AC than that for air-water systems because of higher reversible work consumption in fan. The maximum AC occurs in constant air volume flow system, whereas a minimum value occurs in the chilled ceiling system. The AC of variable air volume flow systemshalf as much as of constant air volume flow system [5,20]. Notably, recent research showed that variable air volume flow system has a great potential of fan energy saving. It is reported that around 0.65kW of electricity is consumed by various auxiliary equipment of HVAC system per ton of cooling load [18].There have been considerable improvement measures to reduce energy consumption of chiller and compressor. In this regard centrifugal chiller efficiency is enhanced by 34% i.e. from CoP (Coefficient of Performance) of 4.24-5.67 (0.83-0.62kW/ton) [21]. The GHG emission results from HVAC system energy consumption are approximately 0.97 kg/kW [22].

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2.   PROBLEM STATEMENT

The function of duct in air distribution system is to transport specific amount of fresh outdoor air, return air, conditioned, supply and exhaust air to or from the conditioned space through space diffusion devices.In order to maintain the specified load of conditioned space and for quiet distribution of air, the velocity of air is limited, sound absorbing material is incorporated and flow obstruction are subsided. The major inefficiency in the air duct operation arises from compression of the insulation material wrapped around it for reducing the heat gain by the conditioned air. In this paper, energy analysis is performed on an HVAC air duct system, working in Novartis pharmaceutical company, Pakistan. It is clear from Fig. 1 that compression of thermal insulation reduces thickness of insulation at corner, bracing and joint of the duct. Thus, it enhances heat transfer to/from the duct.

3.   METHODOLOGY

In order to estimate the effects of compression of thermal insulation at selected points of the duct,an HVAC system ofthe pharmaceutical company is chosen. Firstly, the preliminary data regarding design and operating parameters are collected, which is followed by the development of a mathematical model of air distribution system of the HVAC system using EES software. The model is simulated to analyze the impacts of insulation thickness on heat gain by the duct chilled air. The insulation thickness at the point of compression, duct sheet, mild steel joint and bracing reduce the heat gain byincreasing the value of thermal resistance. In addition to that convective heat transfer coefficient inside and outside the duct reduces heat gain by enhancing convective resistance. Therefore, in this simulation only insulation thickness is varied to estimate its effect on the heat gain, exit temperature, AC and CO_2 emission while others parameters are unchanged.

4.   SELECTED AIR DISTRIBUTON SYSTEM

The schematic view of selected air distribution system for the HVAC system under study is illustrated in Fig. 2. The air distribution system has been divided among five distribution portions i.e. A, B, C, D and E. The portion A having length L_A is connecting the air handling unit with different branches of supply air duct. The conditioned air is distributed to portion B having length L_B and C having

FIG. 1. THERMAL INSULATION COMPRESSION AT SELECTED POINTS OF THE DUCT [23]

length L_C. The portion C supplies conditioned air to Zone-I while conditioned air through B is further divided into two branches D having length L_D and E having length L_E. The branches of portion B supplies air to Zone-II. The cooling load of Zone-II is greater than Zone-I therefore, conditioned air is supplied with two branches in Zone-II and with one in Zone-I.

The design parameters of the duct are illustrated in Fig. 3 and Table 1. The length, width and height of the duct are represented with L, W and H. The thickness of the transverse joint i.e. projected part is t_b and t_t is the horizontal dimension of the joint. The thickness of the galvanized

steel sheet and insulation is represented by t_{sheet} and t_{ins} respectively, whereas thickness of the insulation at selected points of the duct is denoted with t_{comp}. The corresponding design and thermo-physical parameters are given in Tables 1-2 and Fig. 3, respectively [23].

The design and operating parameters of air distribution system are obtained from Novartis pharma Pvt. Ltd. with portable digital hygrometer, portable digital anemometer and pressure transducers as given in Table 2. The value of density, specific heat, kinematic viscosity, Prandtl number and thermal conductivity of conditioned air inside the duct are obtained by using EES.
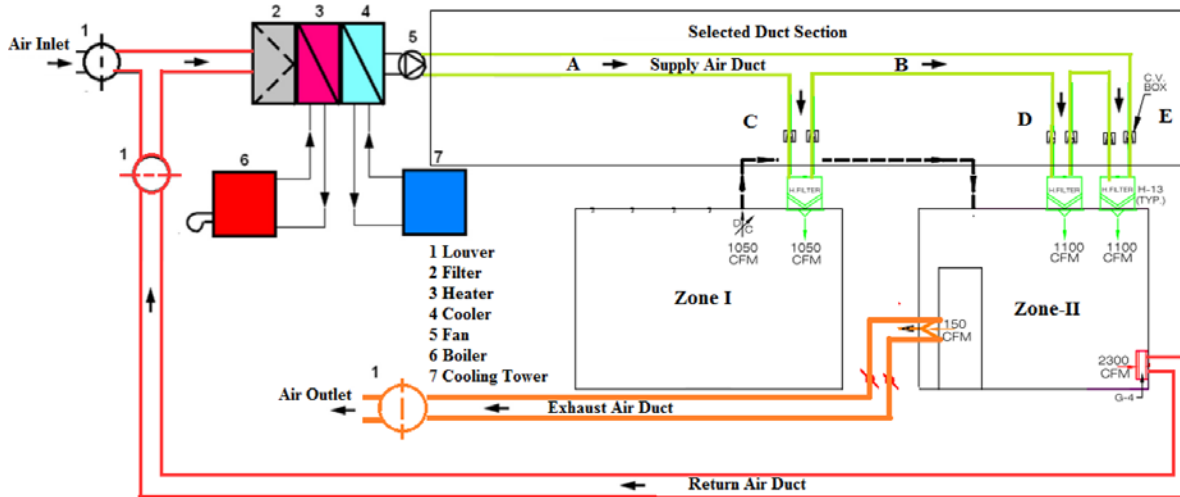

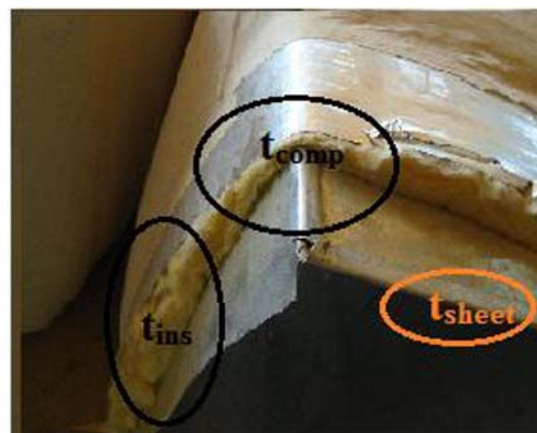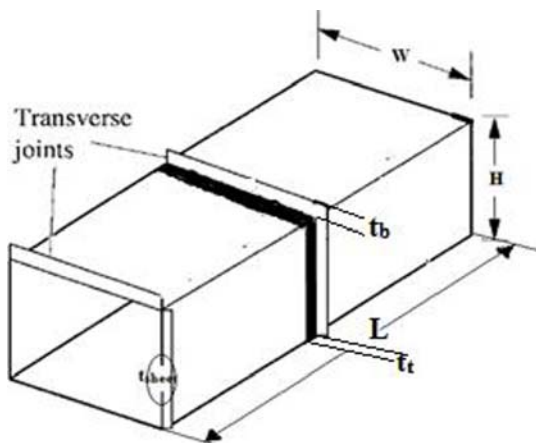FIG. 2. SCHEMATIC LAYOUT OF SIMPLE AIR-DISTRIBUTION SYSTEM [23]

FIG. 3. DESIGN PARAMETERS OF THE DUCT AND INSULATION MATERIAL

5. ASSUMPTIONS

In order to evaluate the performance of the selected air distribution system, a mathematical model has been developed with the following simplified assumptions:

(a)   Steady state conditions are considered.

(b)   Uniform heat gain occurs throughout the duct.

(c)   Temperature of the surrounding of duct is assumed as T_a=303K [23,24].

(d)   Convective heat transfer coefficient of surrounding air is considered as h_o=10 W/m².K [14].

(e)   Thermal conductivity of galvanized steel sheet is 18.18 W/m.K and mild steel angle joint is 54 W/m.K [25].

(f)   The medium pressure duct design has been considered.

(g)   The volume flow rate of the selected air distribution system is around 3250 CFM (Cubic Foot per Minute) and its cooling capacity is 8.76 tons.

6. MATHEMATICAL MODEL

Two basic laws, i.e. conservation of mass and conservation of energy along with different heat transfer concepts have been used to model air distribution system of the HVAC system [2]. These model equations are given below.

Air Mass Flow Rate

$$\dot{m}_A = \dot{m}_{air} = \dot{m}_C + \dot{m}_B$$
$$\dot{m}_B = \dot{m}_D + \dot{m}_E \qquad (1)$$

TABLE 1. DESIGN PARAMETERS OF THE DUCT

Duct Dimensions									
Notation	Width	W(m)	Height	Thickness					Schedule No.
				t_{sheet} (mm)	t_{ins} (mm)	t_{comp} (mm)	t_b (mm)	t_t (mm)	
A	1.2	0.3	1.38	8.5	38	19	3	4	6
B	1.1	0.3	0.31	8.5	38	19	3	4	2
C	0.4	0.3	17.22	7.0	38	19	3	2.5	51
D	0.4	0.3	11.43	7.0	38	19	3	2.5	34
E	0.4	0.3	13.41	7.0	38	19	3	2.5	4

TABLE 2. DESIGN AND OPERATING PARAMETERS

Portion	Mass ($\dot{m}$) kg/s	Average Pressure (P_{avg}) (kPa)	Average Temperature (T_{avg}) K	Specific Heat (ρ) (m³/kg)	Specific Heat (c_p) (kJ/kg.K)	Kinematic Viscosity (υ) (cm2/sec)	Prandtl Number (Pr)(-)	Thermal Conductivity	
								K_{sa} (W/m K)	K_{ins} (W/m K)
A	1.531	102.32	293	1.652	1.014	0.1105	0.7293	0.0251	0.0376
B	1.038	102.22	293	1.650	1.014	0.1106	0.7293	0.0251	0.0376
C	0.495	101.81	294	1.649	1.016	0.1107	0.7292	0.0252	0.0377
D	0.519	101.43	294	1.644	1.015	0.1114	0.7290	0.0252	0.0379
E	0.519	101.46	294	1.644	1.015	0.1114	0.7290	0.0252	0.0379

Cooling Loss

$$\dot{Q}_T = \dot{m}_A(Z_e - Z_i)_A + \dot{m}_B(Z_e - Z_i)_B + \dot{m}_C(Z_e - Z_i)_C + \dot{m}_D(Z_e - Z_i)_D + \dot{m}_E(Z_e - Z_i)_E \quad (2)$$

Heat Gain [26,27]

$$\dot{Q} = UA_s(T_a - Z_{avg}) \quad (3)$$

overall heat transfer coefficient [26,27]

$$U = \sum \frac{1}{R} \quad (4)$$

Thermal resistance without insulation compression

$$R_1 = y\left(\frac{\ln\left(\frac{r_3}{r_2}\right)}{2\pi L_s K_{ins}} + \frac{1}{h_o A_{s,outer}}\right) \quad (5)$$

Thermal resistance at corner

$$R_2 = z\left(\frac{\ln\left(\frac{r_{3,C}}{r_2}\right)}{2\pi L_s K_{ins}} + \frac{1}{h_o A_{s,outer}}\right) \quad (6)$$

Surface without insulation compression

$$y = \frac{2\pi(r_h + t_{sheet})}{2\pi(r_h + t_{sheet} + t_{comp})} \quad (7)$$

Surface insulation compression at corner

$$z = \frac{2\pi t_{comp}}{2\pi(r_h + t_{sheet} + t_{comp})} \quad (8)$$

Thermal resistance without bracing and tip

$$R_s = \left(\frac{1}{h_1 A_{inner}} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi L_s K_{sheet}}\right) + \frac{R_1 R_2}{R_1 + R_2} \quad (9)$$

Thermal resistance at bracing

$$R_b = \left(\frac{1}{h_1 A_{inner}} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi L_b K_{sheet}} + \frac{\ln\left(\frac{r_{3,t}}{r_2}\right)}{2\pi L_b K_b} + \frac{\ln\left(\frac{r_{4,t}}{r_{3,b}}\right)}{2\pi L_b K_{ins}} + \frac{1}{A_o A_{b,outer}}\right) \quad (10)$$

Thermal resistance at tip

$$R_t = \left(\frac{1}{h_1 A_{inner}} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi L_t K_{sheet}} + \frac{\ln\left(\frac{r_{3,t}}{r_2}\right)}{2\pi L_t K_t} + \frac{\ln\left(\frac{r_{4,t}}{r_{3,b}}\right)}{2\pi L_t K_{ins}} + \frac{1}{A_o A_{t,outer}}\right) \quad (11)$$

Total thermal resistance[29]

$$\sum \frac{1}{R} = \frac{1}{R_s} + \frac{1}{R_b} + \frac{1}{R_t} \quad (12)$$

Convective heat transfer coefficient of conditioned air [26-27]

$$h_i = \frac{K_{sa} Nu}{D_h} \quad (13)$$

Hydraulic diameter [26-27]

$$D_h = \frac{4A_c}{p} \quad (14)$$

Reynolds Number [26-27]

$$Re = \frac{V_{air} D_h}{\vartheta} \quad (15)$$

Nusselt number (Laminar) [26-27]

$$Nu = 4.31 \quad (16)$$

Nusselt number (turbulent flow)

$$Nu = \frac{\left(F/8\right)(Re - 1000)Pr}{\left(1 + 12.7\left(Pr^{2/3} - 1\right)\sqrt{F/8}\right)} \quad (17)$$

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Friction factor of conditioned air

$$\frac{1}{\sqrt{F}} = -2.0\log\left(\frac{\varepsilon/D_h}{3.7} + \frac{1}{Re\sqrt{F}}\right) \quad (18)$$

Roughness of duct surface

$$\varepsilon = \left(\frac{1.5\times10^{-4}}{D_h}\right) \quad (19)$$

PCL (Percentage of Cooling Loss)

$$PCL = \left(\frac{\dot{Q}}{Capacity}\right)*100 \quad (20)$$

Auxiliary Consumption

$$W_{aux} = \mu_{aux} * \dot{Q} \quad (21)$$

GHG emissions [22]

$$\dot{m}_{CO_2} = \mu_{CO_2} * \dot{Q}$$
$$\mu_{CO_2} = 0.97 \frac{kgCO}{kW} \quad (22)$$

7. MODEL VALIDATION

The model is validated comparing the simulation results of heat gain with the measured quantities of heat gain, which is obtained from measured values of dry bulb temperature and relative humidity. The dry bulb temperature at the inlet and exit of the duct is measured with temperature gauge. The relative humidity is measured using a digital portable hygrometer. The enthalpy of is obtained using EES property function for mixture of air and water corresponding to measured value psychrometric properties. The measured psychometric properties are given in Table 3.

Fig. 4 exhibits a comparison of simulation and measured quantities of heat gain. The measured value of heat gain is obtained using the values of Table 3 in Equation (2) as shown in Fig. 4. The results show that the heat gain estimated from the developed mathematical model was around 4.29kW while that obtained from the measured values was around 4.52kW. This shows an average difference of less than 5.27% between the measured and model results.

TABLE 3. PSYCHROMETRIC PROPERTIES OF CONDITIONED AIR AT THE INLET AND EXIT OF THE HVAC DUCT

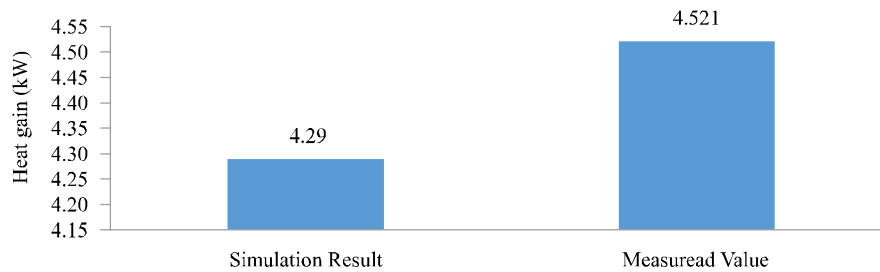| Portion	Temperature (°C)		Relative Humidity (%)		Enthalpy (kJ/kg)	
	$T_{db,i}$	$T_{db,e}$	Φ_i	Φ_o	Z_i	Z_e
A	18.00	18.08	45.23	45.15	32.51	32.67
B	18.08	18.10	45.15	44.26	32.67	32.71
C	18.08	20.31	45.14	41.25	32.67	36.14
D	18.10	20.20	44.26	40.15	32.71	35.03
E	18.10	20.24	44.26	40.23	32.71	35.24

FIG. 4. COMPARISON BETWEEN SIMULATED AND MEASURED QUANTITY OF HEAT GAIN

8.   RESULTS AND DISCUSSION

The effects of compression of thermal insulation on heat gain in different portions of SAD (Supply Air Duct), AC of HVAC system, GHG emission and supply air temperature are illustrated in Figs. 5-8.

8.1   Effect of Insulation Compression on Heat Gain in Different Portions of SAD

Fig. 5 exhibits the effects of insulation thickness on heat gain in different sections of SAD. It is clear from Fig. 5 that increment in the insulation thickness decreases the heat gain.In particular, the heat transfer through duct portions C, D and E isaffected due to smaller cross-sectional area but higher surface area. It is also evident that the heat gain reduces significantlyas the thickness of thermal insulation reduces to 40mm, afterword's effect is insignificant. The maximum heat gain is through section C that is around 1645W and minimum heat gain occurs in section B that is around 37.09W because section C has higher surface area due to larger length and smaller cross-sectional area as compared to section B. However, sections D and E are of same cross sectional area but heat gain in section E is higher since the surface area of portion E is larger. The heat gain decreases from 214-130W in section A, 37.09-21.5W in section B, 1645-973.5W

in section C, 1103-618.5W in section D and 1290-722.9W in section E. The total heat gain in all portions of SAD is reduced from 4.29-2.46kW increasing insulation thickness from 10-40mm at the point of compression.

8.2   Effect of Insulation Compression on AC of HVAC System

The effect of compression of thermal insulation at selected points of the duct on auxiliary power consumption of HVAC system is demonstrated in Fig. 6, which indicates that auxiliary power consumption decreases as insulation thickness increases. The HVAC system under study has the cooling capacity of 8.76 tons, whereas it is estimated by that nearly 0.65 kW of auxiliary power needed for one ton refrigeration.

Machine used as an HVAC system [18]. Correspondingly, the total AC of the system under study is around 5.7 kW and total heat gain by the conditioned air is estimated to be 4.29kW at insulation thickness of 10 mm. Thus, the AC corresponding to its cooling loss due to compression of thermal insulation is 0.80kW/ton capacity. The consumption decreases as the insulation thickness increases, the diminution is significant during initial rise in the thickness. The AC may reduce to 0.46kW at 40 mm thickness, which is around 5% of the total AC of HVAC system.
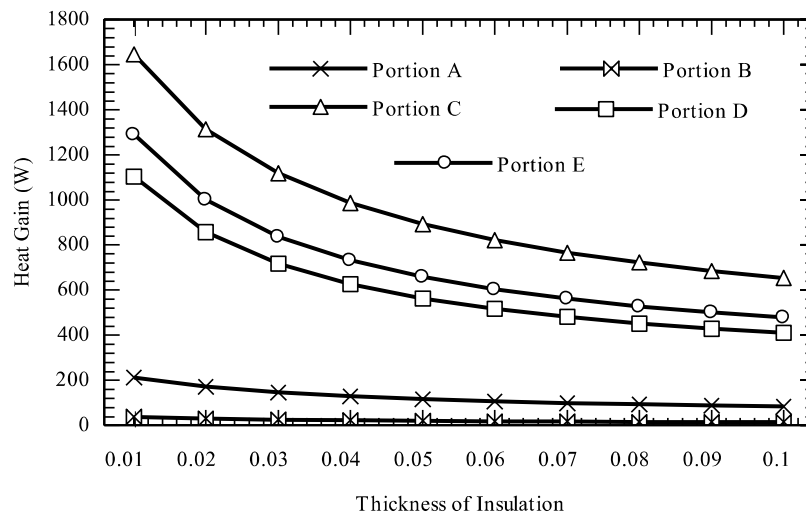

FIG. 5. EFFECTS OF INSULATION THICKNESS ON DIFFERENT PORTIONS OF SUPPLY AIR DUCT

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8.3 Effect of Insulation Compression on CO_2 Emission

The GHG (CO_2) emission varies inversely with insulation thickness as demonstrated in Fig. 7. The variation is more significant during initial rise in the thickness. The CO_2 emission reduces from 4.2-2.3kg/kW by increasing the insulation thickness from 10-40mm at point of compression.

8.4 Effect of Insulation Compression on Dispensed Air Temperature

Fig. 8 exhibits variation in dispensed air temperature in different portions of the duct with respect to insulation thickness. According to Fig. 8 temperature decreases with an increase in the thickness in each portion due to reduction in the heat gain by conditioned air in the ducts. The reduction in temperature is significant in portions C, D and E since these portions have low air flow rates but great surface area. The trend in temperature variation for these portions is similar but highest temperature occurs in portion C which is due to the reason that it has largest surface exposure to the surrounding air. The heat gain into portions A and B is already low; therefore, impact of insulation thickness for these portions of the duct is insignificant.
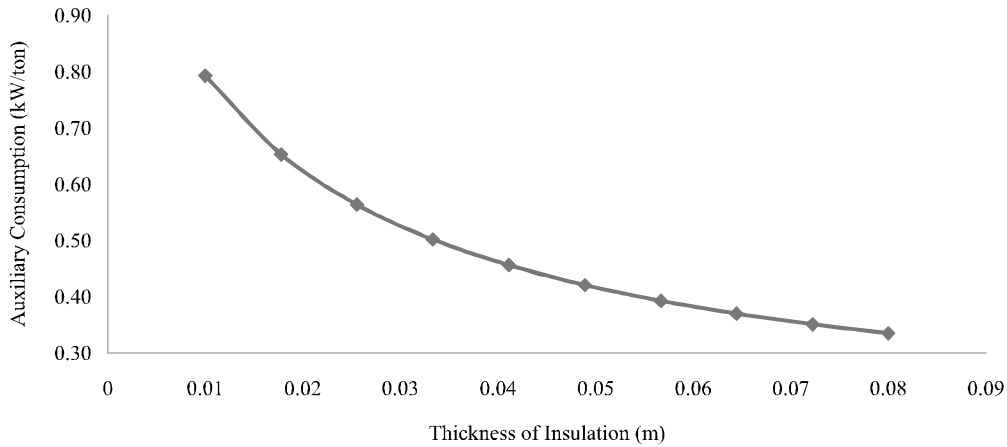

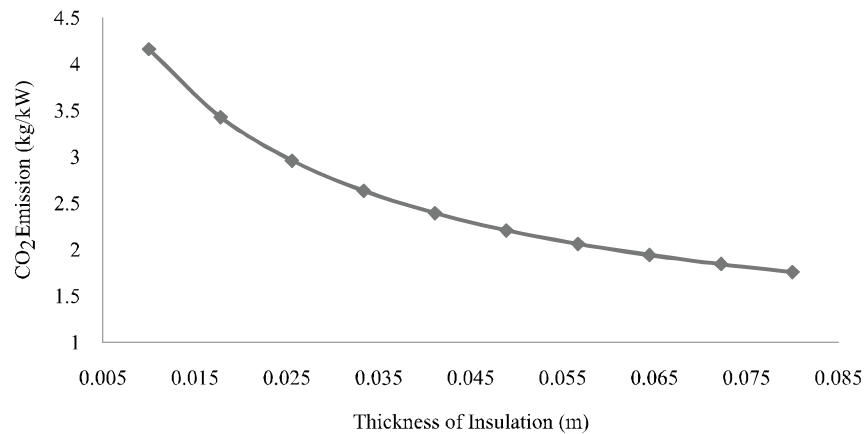*FIG. 6. EFFECT OF THERMAL INSULATION THICKNESS ON AUXILIARY CONSUMPTION*

FIG. 7. EFFECTS OF COMPRESSION OF THERMAL INSULATION ON CO_2 EMISSIONS

Mehran University Research Journal of Engineering & Technology, Volume 36, No. 3, July, 2017 [p-ISSN: 0254-7821, e-ISSN: 2413-7219]
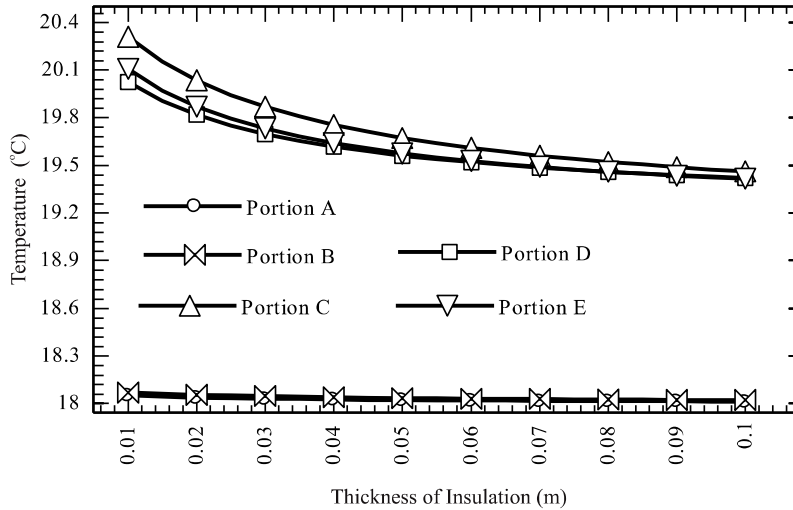
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FIG. 8. EFFECT OF INSULATION THICKNESS ON DISPENSED AIR TEMPERATURE IN DIFFERENT PORTIONS

9. CONCLUSIONS

Following results are concluded from this study:

(i) The developed mathematical model is validated with cooling loss estimated using psychometric parameters of conditioned air with an error of 5.27%.

(ii) The total heat gain from surrounding into the duct is reduced from 4.29-2.46kW by increasing insulation thickness from 10-40mm at selected points of the duct.

(iii) The energy loss due to heat gain in selected air distribution system could be decreased from 14-8% with optimum thickness i.e. around 6% of cooling load could be saved.

(iv) The AC due to cooling loss in the selected air distribution system can be reduced from 0.80-0.46kW.

(v) The GHG emission can be reduced from 4.2-2.3kg/kW by increasing thickness of insulation from 10-40mm.

(vi) The temperature of conditioned air can be decreased up to 0.4°C.

(vii) An increase in the thickness of thermal insulation at the selected points of the duct increases the quantity of thermal insulation by 12%.

10. NOMANCLATURE

A	=	Duct's external surface area (m^2)
D_h	=	Hydraulic Diameter of the duct (m)
H	=	Height of the duct (m)
h	=	Convective heat transfer coefficient (W/m^2.K)
L	=	Length of the Duct (m)
$\dot{m}$	=	Mass Flow Rate (kg/s)
n	=	Schedule
Nu	=	Nusselt Number (-)
p	=	Pressure (kPa)
Pr	=	Prandtl Number (-)
$\dot{Q}$	=	Rate of Heat gain or Loss (W)
$\dot{Q}_T$	=	total cooling loss (W)
Re	=	Reynolds Number (-)
r	=	Radius (m)
T	=	Temperature(°C)

t	=	Thickness (m)
U	=	Overall Heat Transfer Coefficient (W/m^2.K)
$\dot{V}$	=	Volume Flow Rate (m^3/s)
V	=	Velocity (m/s)
W	=	Width of the Duct (m)
y	=	the proportion of the uncompressed surface area to total surface area of the duct without tip and bracing (-)
z	=	represents the proportion of the compression surface area of corner to total surface area of the duct (-)
Z	=	Enthalpy of conditioned air (kJ/kg)

Greek Letters

R	=	Thermal resistance (K/W)
U	=	overall heat transfer coefficient of the layers(W/m^2.K)
K	=	Thermal conductivity (W/m.K)
ϑ	=	Kinematic viscosity (m^2/s)
F	=	friction factor (-)
ε	=	relative roughness (-)
μ	=	Conversion Factor (kg/kW)

Abbreviations

AC	Auxiliary Consumption
CFM	Cubic Foot per Minute
EES	Engineering Equation Solver
PCL	Percentage of Cooling Loss

Subscripts

a	=	ambient
avg	=	average
b	=	bracing
comp	=	compression
e	=	exit
h	=	hydraulic
i	=	inlet
ins	=	insulation
psy	=	psychometric
s	=	surface
sa	=	supply air
t	=	tip

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